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Thermodynamic analysis of a Rankine cycle applied on a diesel truck engine using steam and organic medium

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ABSTRACT

A theoretical study is conducted to investigate the potential improvement of the overall efficiency of a heavy-duty truck diesel engine equipped with a Rankine bottoming cycle for recovering heat from the exhaust gas. To this scope, a newly developed thermodynamic simulation model has been used, considering two different working media: water and the refrigerant R245ca. As revealed from the analysis, due to the variation of exhaust gas temperature with engine load it is necessary to modify the Rankine cycle parameters i.e. high pressure and superheated vapor temperature. For this reason, a new calculation procedure is applied for the estimation of the optimum Rankine cycle parameters at each operating condition. The calculation algorithm is conducted by taking certain design criteria into account, such as the exhaust gas heat exchanger size and its pinch point requirement. From the comparative evaluation between the two working media examined, using the optimum configuration of the cycle for each operating condition, it has been revealed that the brake specific fuel consumption improvement ranges from 10.2% (at 25% engine load) to 8.5% (at 100% engine load) for R245ca and 6.1% (at 25% engine load) to 7.5% (at 100% engine load) for water.

1. Introduction

Greenhouse effect and depleted petroleum supplies are crucial issues that the developed world's economies are facing. Thus, governments in the industrialized countries have introduced strict regulations for diesel engine emissions and fuel economy standards. The last two decades, diesel engine manufacturers have improved significantly both fuel consumption and engine efficiency by applying various technologies. However, there are perspectives for further improvement of specific fuel consumption when recovering the exhaust gas heat considering that diesel engines still reject a major part of fuel chemical energy to the environment through exhaust gases.

In the early 1970s a research program funded by US Department of Energy (DOE) was conducted by Mack Trucks and Thermo Electron Corporation [1–3]. During this program, an Organic Rankine Cycle System (ORCS) was installed on a Mack Truck diesel engine. The lab test results revealed an improvement of brake specific fuel consumption (bsfc) of 10–12.7%. These percentages were verified from highway tests. Argonne National Laboratory [4] considered an Integrated Rankine Bottoming Cycle (IRBC) using the engine coolant as working media. Finally, Thermo Electron Corporation [5] conducted an investigation using an advanced organic fluid having a high thermal and chemical stability. This investigation also revealed a significant potential for exhaust heat recovery.

Systems developed today differ from those of the 1970s because of the advances in the development of expansion devices and the broader choice of working fluids. The literature survey indicated that, at the present time, Rankine cycle systems are under development, but no commercial solution seems to be available yet. Most of the systems recover heat from the exhaust gas [6–8] and, in addition from the cooling circuit. The control of the system is particularly complex due to the (often) transient regime of the heat source. However, optimizing the control is crucial to improve the performance of the system. For instance, Honda [6] proposed to control the temperature by varying the water flow rate through the evaporator (by varying the pump speed) and to control the expander supply pressure by varying its rotational speed.

In the present work, an analysis is conducted to evaluate the potential for heat recovery from a heavy duty DI diesel truck engine using a Rankine cycle system [9–11]. Initially, H_2O has been considered as Rankine cycle working medium. Then the organic medium R245ca (1,1,2,2,3-pentafluoropropane), has been examined, which belongs to the family of fluorocarbon compounds. The selection of this type of refrigerant has been made because it is

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Nomenclature

A	heat transfer area (m^2)
A_{tot}	total heat transfer area (m^2)
c_p	heat capacity under constant pressure (J/kg K)
D_e	equivalent diameter (m)
D	diameter (m)
f	friction factor (–)
F	correction factor (–)
G_h	exhaust gas mass flow rate per area unit (kg/m^2 s)
H	heat transfer coefficient (W/m^2 K)
h	enthalpy (J/kg)
k	thermal conductivity (W/mK)
L	heat exchanger length (m)
$\dot{m}$	mass flow rate (kg/s)
Eff	efficiency (–)
Nu	Nusselt number (–)
N_b	Baffles number (–)
N_p	number of tube passes (–)
$\dot{Q}$	heat (W)
P	pressure (Pa)
Power_{pump}	power consumption of circulator pump (W)
Pr	Prandtl number (–)
PT	tube pitch (distance between successive tubes) (–)
R_{ws}	thermal resistance of tube material (K/W)
Re	Reynolds number (–)
T	temperature (°C)
U_0	total heat transfer coefficient (W/m^2K)
u_{c_m}	mean velocity of fluid at the cold side (m/s^2)
Δp_{total}	total pressure drop at the tube side (Pa)
Δp_s	pressure drop at the shell side (Pa)
$\Delta T_{lm,cf}$	logarithmic mean temperature difference (°C)

Special characters

μ	viscosity (Pa s)
ρ	density (kg/m^3)
φ	correction factor for fluid viscosity at the hot side (–)
λ	lambda ratio (–)

Subscripts

c	fluid at the cold side of heat exchanger
CAC	charge air cooler
exp	Rankine cycle expander
h	fluid at the hot side of heat exchanger
H	higher value
i	tube inside
in	inlet
is	isentropic process
L	lower value
o	tube outside
out	outlet
RC	Rankine cycle
s	heat exchanger shell
wf	Rankine cycle working fluid
ws	tube material
wfin_exp	expander inlet
wfout_exp	expander outlet
wfin_pump	pump inlet

Abbreviations

bsfc	brake specific fuel consumption
EGR	exhaust gas recirculation
SCR	selective catalytic reduction
DPF	diesel particulate filter

inflammable, non toxic and ozone friendly, presenting zero ozone depletion potential (ODP) and low global warming potential (GWP) [12].

A typical layout of this technology is depicted in Fig. 1, where an exhaust gas heat exchanger is installed downstream of the turbocharger turbine providing the heat required by the working medium of the Rankine cycle. A circulator-pump and an expander coupled with an electric generator are also included in the system. The installation of a radiator is vital for the proper operation of the system considering that the expanded vapor rejects heat to the ambience during its condensation. Another major component as depicted in this figure is the Exhaust Gas Recirculation (EGR) cooler (where additional exhaust heat recovery is achieved).

The operation of the Steam Rankine cycle is based on the following thermodynamic processes. The working medium enters the circulation pump at its saturated liquid state and exits at high pressure p_H. The installation of a heat recuperator improves the thermodynamic efficiency of Rankine cycle and reduces the dimensions of the condenser. Following this, the working medium is preheated to the saturated liquid state, when it flows into the exhaust gas heat exchanger. The required thermal energy for working medium evaporation and superheating is covered by the main exhaust gas stream and partially by the exhaust gases flowing through the EGR cooler. Utilization of EGR heat is favorable for cycle efficiency because of its superior thermodynamic characteristics (i.e. higher temperature level). The superheated vapor expands from the high pressure p_H to low pressure p_L. However, when the expanded vapor is nearly saturated then the inclusion of a heat recuperator in Rankine cycle system is not feasible. This is why the recuperator is used when the working medium is R245ca.

For each operating point of the diesel engine, the optimum Rankine cycle operating conditions (defined by the working medium mass flow rate and cycle high pressure) are estimated to maximize generated power. All the relevant restrictions regarding the mechanism of heat transfer have been considered in the analysis for the main exhaust gas stream and EGR heat exchangers (i.e. pinch point requirements). The same restrictions have been taken into account for the heat recuperator.

The main purpose of the present investigation is to evaluate the effect of heat recuperation from the EGR and the main exhaust gas stream on engine performance. Thus, the analysis is based on the derived results for bsfc improvement, generated power, working fluid mass flow rate, cycle high pressure and cycle efficiency. Using

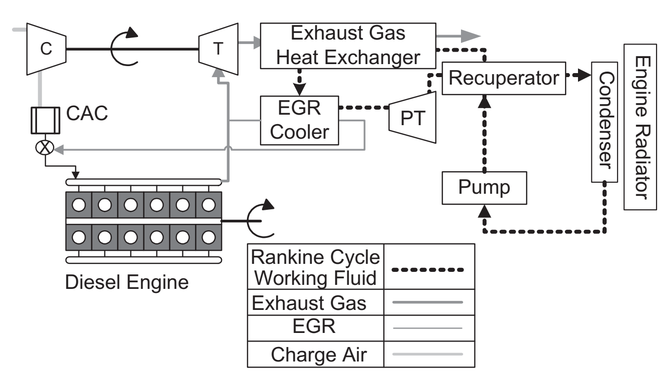

Fig. 1. Schematic view of rankine cycle system installed in a diesel engine.

the in-house simulation model, all the components of the engine layout are fundamentally described, and it is possible to conduct parametric studies varying the engine operating conditions, the working medium and the characteristics of the Rankine cycle (high pressure, mass flow rate, etc.), taking into consideration specific design criteria (i.e. pitch point requirement) and determine the optimum design for each operating condition. Through the analysis conducted it has been revealed that it is necessary to control the high pressure of the Rankine cycle to achieve optimum performance at all engine loads. Moreover, using the simulation model the effect of the Rankine cycle on the engine cooling system is revealed and using as parameter the high pressure of the Rankine cycle, it is possible to control the loading of the engine cooling system.

The analysis reveals that Rankine cycle system provides potentiality for heat recovery from several heat sources of the diesel engine. Considering increasing fuel prices and crucial environmental issues (global heating) it appears that Rankine cycle could be possibly used as an exhaust gas heat recovery technique to achieve further reduction of engine bsfc and curtail CO_2 specific emissions of diesel engines.

2. Rankine cycle simulation model

A detailed simulation model has been developed to evaluate in which extend the addition of a Rankine cycle system is beneficial for recovering exhaust gas heat from a diesel engine. The required thermodynamic and transport properties for the working media examined (R245ca and H_2O) are obtained from the electronic database REFPROP [13]. The exhaust gas properties are calculated using polynomial expressions [14] and the exhaust gas composition is estimated from lambda ratio.

Since the examined diesel engine is equipped with an EGR system, the heat recovery from the main and the recirculated exhaust gas streams (EGR) are both taken into account. The type of heat exchanger used for heat recuperation from exhaust gas and EGR stream is the shell and tube type with two tube passes, which is a conventional solution, especially for the high pressures and temperatures existing in the working cycles considered. When Rankine cycle operates with R245ca as working medium, a heat recuperator is used to preheat the organic fluid after the circulation pump, utilizing the excess heat of the working medium after being expanded in the power turbine of the Rankine cycle. It should be mentioned that in the present study, is not taken into account the heat that could be utilized by the engine coolant or the intercooler, mainly due to the lack of required data. Furthermore, the low temperature of the engine coolant is a serious obstacle for using it in Rankine cycle system. However, the engine coolant heat could cover a small part of preheating the organic working medium.

Rankine cycle high pressure p_H is a critical parameter of the analysis. To determine its optimum value for each case, examined simulation runs have been iteratively conducted by varying p_H between a minimum and a maximum value with an incremental step equal to 1 bar. The second parameter investigated in the present study is the superheated vapor temperature at the expander inlet. The minimum value of this temperature must fulfil the requirement that the working medium after the expansion is at a superheated vapor condition to avoid condensation inside the turbine. This value is obviously affected from the high pressure p_H of the working cycle. The maximum value of the temperature at the expander inlet is provided from the electronic database REFPROP. For this parametric investigation, an incremental step equal to 0.1 °C is used to determine the optimum working fluid temperature at the expander inlet.

In the following paragraphs is given a description of the calculation procedure for Rankine cycle components.

2.1. Performance of exhaust gas and EGR heat exchanger

The simulation model divides the exhaust gas heat exchanger into three different components: preheater, evaporator and superheater.

The total extracted heat amount from the main exhaust gas and EGR gas streams is given in the following relation:

$$\dot{Q} = \dot{m}_h Cp_h (T_{h,in} - T_{h,out}) = \dot{m}_{wf} (h_{c,out} - h_{c,in}) \tag{1}$$

Obviously, to secure heat transfer from the hot to the cold side of the heat exchanger, the temperature of exhaust gas must be always higher than the corresponding one of working fluid during the entire process. For this reason, the value of the exhaust gas temperature after each part of the heat exchanger must comply with the pinch point requirement, which is described by the following expression:

$$T_{h,out} \geqslant T_{c,in} + 10\ ^\circ\mathrm{C} \tag{2}$$

In the present analysis, the simulation of the heat exchanger is based on the Logarithmic Temperature Mean Difference (LTMD) method. Therefore, the heat exchange area at each component part of the heat exchanger is estimated from the following expression [15]:

$$A_0 = \frac{\dot{Q}}{U_0 F \Delta T_{lm,cf}} \tag{3}$$

where correction factor F is calculated from charts [15] corresponding to a counter flow shell and tube heat exchanger with even number of tube passes.

The calculation of working fluid mass flow rate, the exhaust gas and EGR gas temperature variation, is based on the aforementioned expressions for exchanged heat amounts. The logarithmic mean temperature difference $\Delta T_{lm,cf}$ at the heat exchanger is estimated from the inlet and outlet temperature values of the hot and cold side of the heat exchanger as follows [15]:

$$\Delta T_{lm,cf} = \frac{(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})}{\ln\left(\frac{T_{h,in} - T_{c,out}}{T_{h,out} - T_{c,in}}\right)} \tag{4}$$

The calculation procedure is based on the expressions for energy balance (1) and heat exchange area (3). The simulation model initiates with the estimation of exhaust gas temperature at the exit of the EGR heat exchanger and the working fluid mass flow rate and then continues with the calculation of exhaust gas temperatures before and after the evaporator and the preheater. During this procedure it should be satisfied the following constraint: The estimated total heat transfer area should not exceed the corresponding limit of the exhaust gas heat exchanger.

Moreover it should be stated that the exhaust gas heat exchanger should be placed after DPF catalyst to reduce the risk of fouling. The specific heat exchanger could be installed before the SCR catalyst to control the inlet temperature. In this case exhaust gas cooling can protect the SCR catalyst especially during active filter regeneration, where the exhaust gas temperature is extremely high. Temperature management is also feasible by cooling partially the exhaust gas in order to improve the efficiency of SCR.

2.2. Simulation of the heat exchangers

As already mentioned, both heat exchangers used for the exploitation of exhaust gas and EGR heat belong to the shell and tube type, while the recuperator is of the finned tube type 8.0–3/8 T. In the next paragraphs is given a brief description of the procedure followed to calculate the heat exchange coefficient and the pressure drop at both sides of the heat exchanger.

2.2.1. Shell and tube heat exchanger

2.2.1.1. Heat transfer coefficients. The overall heat transfer coefficient U_0, based on the outside diameter of the tubes is calculated according to the following relation [16,17]:

$$\frac{1}{U_0} = \frac{1}{H_i} + A_0 R_{ws} + \frac{1}{H_0} \tag{5}$$

The shell-side heat transfer coefficient H_0 is estimated using the McAdams [11,12] correlation as follows:

$$\frac{H_0 D_e}{k_h} = 0.36 Re_s^{0.55} \left(\frac{c_{ph}\mu_h}{k_h}\right)^{1/3} \left(\frac{\mu_h}{\mu_{ws}}\right)^{0.14} \tag{6}$$

The tube side heat transfer coefficient H_i is calculated from the relation:

$$H_i = \frac{Nu_c k_c}{D_i} \tag{7}$$

When Reynolds number Re_c is lower than 1×10^4 then, the Nusselt number is given by (8) [16,17]:

$$Nu_c = 0.023\ Re_c^{0.8}\ Pr_c^{0.8} \tag{8}$$

For higher values of Reynolds number ($Re_c > 1 \times 10^4$), the Nusselt number is provided from the Gnielinski's correlation [16,17], as shown in (9).

$$Nu_c = \frac{\left(\frac{f_c}{2}\right)(Re_c - 1000)Pr_c}{1 + 12.7\left(\frac{f_c}{2}\right)^{\frac{1}{2}}\left(Pr_c^{\frac{2}{3}} - 1\right)} \tag{9}$$

where factor f_c is a logarithmic function of the Reynolds number:

$$f_c = (1.58 \ln Re_c - 3.28)^{-2} \tag{10}$$

The ratio of thermal resistance to the heat flow via conduction through the tubes is given by (11)

$$R_{ws} = \frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi Lk} \tag{11}$$

where L denotes tube length and k is the thermal conductivity of the tube material. In the present study the tube is assumed to be made of Aluminum, with $k = 237$ W/mK.

2.2.1.2. Pressure drop. The pressure drop on the shell of the exhaust gas heat exchanger is calculated using the following equation [11,12]:

$$\Delta p_s = \frac{f_s G_h^2 (N_b + 1) D_s}{2\rho_h D_e \phi_s} \tag{12}$$

While the friction factor f_s for the shell is calculated from [16]:

$$f_s = \exp(0.576 - 0.19 \ln Re_s) \tag{13}$$

Then the total pressure drop of the tube side is provided by the following expression [16,17]:

$$\Delta p_{total} = 4 f_c \frac{LN_p}{D_i} \frac{\rho_c u_{c\_m}^2}{2} + 4 N_p \frac{\rho_c u_{c\_m}^2}{2} \tag{14}$$

The first term of the previous expression is the pressure drop of the working fluid which flows inside the tubes and the second term corresponds to the pressure drop due to the variation of direction in the tube passes.

2.2.2. Performance of heat recuperator

The finned tube 8.0–3/8 T heat exchanger type is selected for heat recuperator [16,17]. It should be mentioned that the fluid flowing inside the tubes is the cold side of the finned tube heat exchanger. Heat exchange area is estimated in a similar manner to the exhaust gas heat exchanger (3), but the correction factor F is provided from charts corresponding to a cross flow heat exchanger. The heat exchange coefficient at the hot side is provided from the following expression:

$$H_0 = St G_h c_p \tag{15}$$

where the Stanton number (St) is estimated from the Prandtl number and the Colburn parameter according to the expression:

$$J_H = St Pr^{\frac{2}{3}} \tag{16}$$

Colburn parameter is provided from a chart corresponding to the selected finned tube heat exchanger. The exhaust gas mass flow rate per unit area is expressed as follows:

$$G_h = \frac{\dot{m}}{A_{flow\_min}} \tag{17}$$

The minimum flow area is estimated from the frontal area of the heat exchanger considering the characteristic parameter σ provided from the following expression:

$$\sigma = \frac{A_{flow\_min}}{A_{fr}} = 0.534 \tag{18}$$

2.2.2.1. Pressure drop inside heat recuperator. The pressure drop in the outer side of tubes is estimated as follows:

$$\Delta p_h = f \frac{G_h^2}{2\rho} \frac{A_{tot}}{A_{flow\_min}} \tag{19}$$

The friction factor f, is estimated from a corresponding chart for the selected finned tube heat exchanger.

2.3. Power units

2.3.1. Expander

In general positive displacement expanders are more suitable compared to the turbine ones in Rankine cycle applications with low mass flow rate of the working media. Therefore, expander type depends on the working media used and the size of the installation. For automotive applications (small engines), where steam is used, the use of a positive displacement expander is the only solution, due to the low mass flow rate. As far as efficiency is concerned, again the positive displacement expanders appear to be more advantageous at low load conditions. However, difficulties may arise when using a direct coupling to the engine crank shaft due to limitation in rotational speed. At high load conditions, turbine and positive displacement expanders have similar efficiencies which are even over 80% [18].

The superheated steam is expanded from the high pressure to the Rankine cycle low pressure p_L. Thus the generated power from the expander ($Power_{exp}$) is given from:

$$Power_{exp} = \dot{m}_{wf}(h_{wfin\_exp} - h^{is}_{wfout\_exp}) Eff_{is\_exp} \tag{20}$$

where the expander isentropic efficiency Eff_{is_exp} is assumed to be constant and equal to 85% (which is the maximum expected value). Previous research works [18,19] focusing on Rankine cycle expanders proved that the isentropic efficiency of a radial/axial turbine or a piston expander could surpass the value of 80%. The maintenance of constant efficiency is feasible with the variation of the expander speed and pressure ratio.

2.3.2. Circulator pump

The required power to drive the circulator pump is given from the following equation:

Table 1
Engine operating conditions considered for the investigation at 1700 rpm engine speed.

Load (%)	Power (kW)	$\dot{m}_{exh}$ (kg/s)	$\dot{m}_{EGR}$ (kg/s)	λ (–)	P_{EGR} (bar)	T_{exh_in} (°C)	T_{EGR_in} (°C)
100	366.6	0.4945	0.0982	1.54	4.14	397.8	581.0
75	277.8	0.4058	0.1046	1.64	3.87	354.3	518.2
50	183.6	0.2993	0.1314	1.69	3.79	306.6	455.3
25	90.0	0.1784	0.1194	1.78	2.47	285.3	383.4

Table 2
Recuperator, exhaust gas and EGR heat exchanger dimensions.

	Recuperator	Exhaust gas	EGR
Heat exchanger length (m)	0.50	1	0.50
Heat exchanger volume (m^3)	0.08	20.00	10.00
D_i (mm)	8.2	7	7
D_0 (mm)	10.2	8	8

$$\text{Power}_{\text{pump}} = \dot{m}_{\text{wf}} \frac{(P_{\text{H}} - P_{\text{L}})}{\rho_{\text{wfin\_pump}} \text{Eff}_{\text{pump}}} \tag{21}$$

where the adiabatic efficiency Eff_{pump} of the circulator pump is assumed equal to 85%.

3. Test cases examined

The engine considered herein is a six-cylinder heavy-duty two stage turbocharged truck diesel engine having a bore of 125 mm, a stroke of 140 mm and compression ratio equal to 16.5:1. The maximum value of bmep is equal to 33 bar, it is thus a significantly downsized engine. The engine is equipped with a common-rail fuel injection system and a high pressure loop EGR, which means that a part of the exhaust gas before the turbocharger turbine is abstracted and directed to an EGR cooler before entering he engine's inlet manifold.

In the present work the parametric investigation has been conducted at various loads ranging from 25% to 100% at 1700 rpm engine speed, which is the medium operating speed of the engine. The entire set of engine operating conditions considered is presented in Table 1.

The estimated dimensions of the heat recuperator, the exhaust gas and EGR heat exchangers are given in Table 2, while the additionally required data for exhaust gas and EGR heat exchangers are presented in Table 3.

4. Results and discussion

As mentioned, a parametric analysis has been conducted for the case of a Rankine cycle system coupled to the exhaust of a heavy duty DI diesel engine. The main parameter of the analysis is the Rankine cycle high pressure p_H, which is varied between a minimum and a maximum value with an incremental step equal to 1 bar. The reason for this variation is actually attributed to the variation of engine operating conditions that affect the temperature level of the exhaust and thus the heat exchange between the working medium and the exhaust gas. The maximum and minimum values of Rankine cycle high pressure p_H are provided in Table 4 for both working media examined (water and R245ca). In the same

Table 3
Exhaust gas and EGR heat exchanger required data.

PT (mm)	Number of tubes	Number of tube passes (NP)	Baffle spacing B (m)
12	400	2	0.500

Table 4
Rankine cycle high and low pressure for both working media examined.

Working medium	Max. cycle high pressure p_H (bar)	Min. cycle high pressure p_H (bar)	Cycle low pressure p_L (bar)	Condensation temperature T_L (°C)
H_2O	50	5	0.60	85.93
R245ca	36	8	2	44.23

table, the selected values for the lower pressure p_L of the Rankine cycle and the corresponding condensation temperature T_L are depicted.

Ambient temperature is asummed equal to 20 °C. The condensation temperature should be at least 20 °C greater than the ambient temperature to guarantee safe transfer of heat from the Rankine cycle radiator to the ambience. For this reason, the Rankine cycle low pressure is defined by the corresponding value of the condensation temperature. It should be mentioned that as observed in Table 4, the condensation temperature considered when water is used is higher than the one used for R245ca. This is due to the fact that in the present work, the limit of the low condensation pressure has been set to 0.6 bar for steam, to avoid technical difficulties which arise when very low pressures are used. Apart from this, the use of a low condensation pressure of 0.1 bar for steam, to have the same condensation temperature as for the organic medium, would create additional difficulties for the expander due to the high expansion ratio.

The analysis is conducted using the aforementioned simulation model to estimate the operating parameters of the Rankine cycle system. The impact of heat recovery from the main gas stream and the EGR cooler is examined separately for both working media considered. The target of the calculations is the estimation of the potential bsfc improvement, the maximum power output, the rejected heat to the ambience, the absorbed heat from the exhaust gases and the impact of the exhaust gas heat exchanger on engine backpressure (which obviously affects engine net efficiency through its effect on the gas exchange). The optimum Rankine cycle at each engine operating point is determined from the simulation model through the aforementioned variation of cycle high pressure p_H and temperature.

4.1. Effect on overall bsfc

The positive effect of exhaust gas heat recuperation with Rankine cycle system on engine performance is clearly depicted in Fig. 2. Specifically, there is a significant improvement of overall

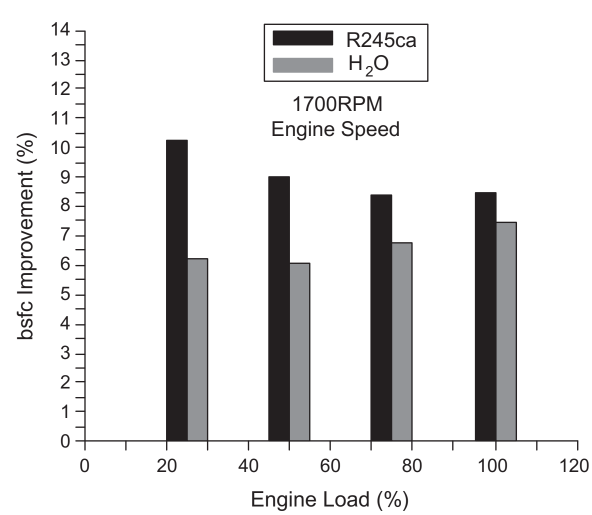

Fig. 2. Bsfc improvement vs engine load when using either steam or R245ca as Rankine cycle working media at 1700 rpm.

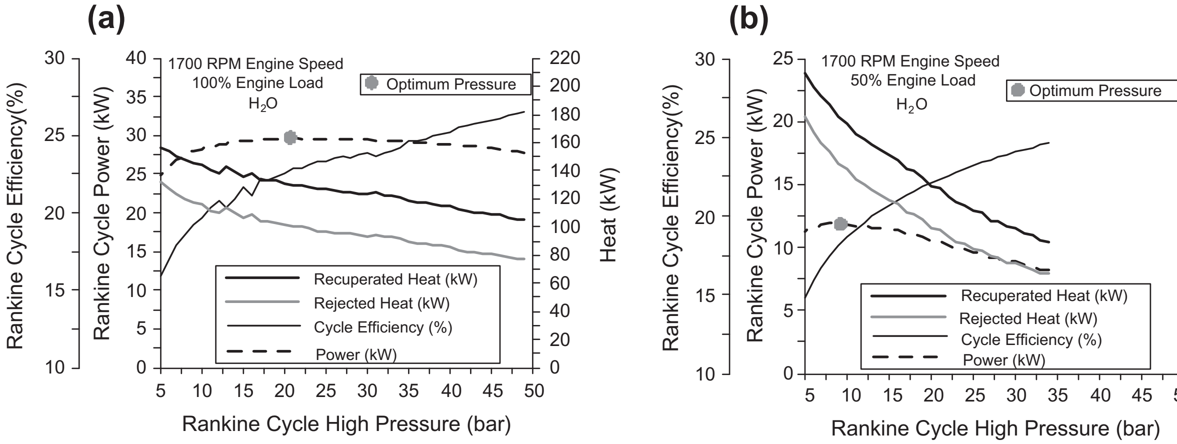

Fig. 3. Thermodynamic efficiency, net generated power, rejected heat and total heat absorbed vs Rankine cycle high pressure using steam as working medium at 1700 rpm, 100% and 50% of full engine load.

bsfc (i.e. fuel consumption/total power) at all engine loads for both working fluids examined (steam or R245ca). However, the improvement is higher for the organic Rankine cycle compared to the steam case, at the entire range of engine loads examined. The depicted values of bsfc improvement at Fig. 2 correspond to the optimum operating point of the Rankine cycle as determined above. Specifically as revealed, the installation of the organic Rankine cycle improves overall bsfc by 8.5%, 8.4%, 9% and 10.2% at 100%, 75%, 50% and 25% of engine load respectively. The corresponding improvement of bsfc for steam Rankine cycle is 7.5%, 6.8%, 6.1% and 6.2% at 100%, 75%, 50% and 25% of engine load. It should be mentioned that one of the characteristics of R245ca is the low vaporization temperature, which makes its use favorable especially at part engine load conditions, where exhaust temperature is low. Due to this, as shown in Fig. 4a and b, the efficiency of the organic Rankine cycle at part load is higher than the one at high load, while for steam (Fig. 3a and b) the opposite is observed. For this reason the replacement of steam with the organic medium R245ca improves further overall bsfc at 25% and 50% engine load, as shown in Fig. 2.

4.2. Effect on rankine cycle efficiency-generated power-total heat absorbed–rejected heat to the ambience

As already mentioned, the main target of the simulation model is to estimate the optimum operating characteristics of the Rankine cycle in order to achieve the maximum efficiency gain. Figs. 3 and 4 provide results for the impact of cycle high pressure p_H on Rankine cycle efficiency, generated output, rejected and total absorbed heat, using either water or R245ca as working medium. These figures also depict the calculation procedure followed by the simulation model.

The results provided in Fig. 3a and b correspond to the use of the steam Rankine cycle at full (Fig. 3a) and part engine load (Fig. 3b). As revealed the net generated power of the Rankine cycle increases with the increase of cycle high pressure and then starts to decline. This effect is more intense at part engine load (Fig. 3b). The value of the high pressure of the Rankine's cycle $p_{H,}$ at which the generated power peaks, is considered as the optimum one.

On the other hand, when R245ca is used as the Rankine cycle working medium, (Fig. 4a and b) it is observed that the generated power at both engine loads examined (part and full) increases asymptotically as the high pressure of the cycle reaches its peak value. Moreover, comparing Figs. 3a and b and 4a and b, it is revealed that for steam the optimum value of the high pressure p_H decreases as engine load decreases, contrary to what happens in the case of the organic medium R245ca, where the optimum p_H remains practically the same with the variation of engine load. This appears to be positive for the organic cycle since less control of P_H will be required.

As far as the Rankine cycle efficiency is concerned, Figs. 3a and b and 4a and b reveal that there is an improvement when cycle high

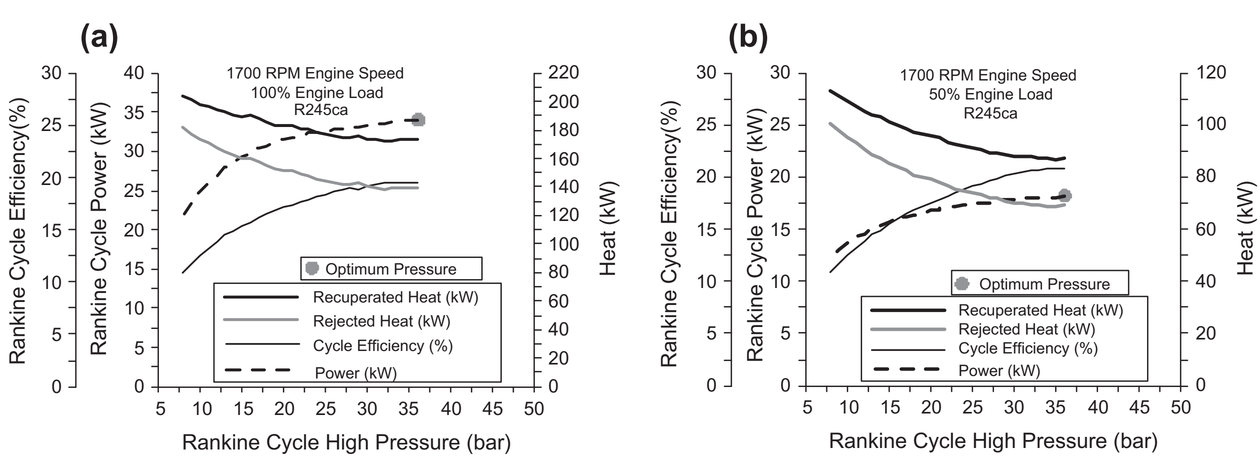

Fig. 4. Thermodynamic efficiency, net generated power, rejected heat and total heat absorbed vs Rankine cycle high pressure using R245ca as working medium at 1700 rpm, 100% and 50% of full engine load.

pressure increases as expected. As a consequence, less amount of heat is absorbed from the exhaust gas for the same generated power. Therefore at the same time the increase of Rankine cycle high pressure, leads to lower amounts of rejected heat to the ambience. The reduction of engine cooling system demands is of vital importance for the application of the examined heat recovery technique on a heavy duty truck diesel engine, where packaging difficulties exist.

Thus, for actual engine applications, the Rankine cycle should operate at the highest possible vaporization pressure values especially at full engine load conditions, where the cooling demands are higher. As witnessed from Figs. 3a and b and 4a and b, the reduction of rejected heat as vaporization pressure increases has a slight impact on the maximum generated power using either steam or R245ca as working medium.

The comparison of both working media examined reveals that the steam Rankine cycle presents lower values of generated power, absorbed and rejected heat compared to the organic. However, the last operates with lower values of thermodynamic cycle efficiency. However at the end, the maximum generated power is 34 kW and 30 kW at 100% load when using R245ca and steam respectively.

4.3. Effect on the optimum working fluid mass flow rate

The examined heat recovery system generates its maximum power for a certain value of working medium mass flow rate. The histogram in Fig. 5 provides the optimum value of steam and R245ca mass flow rates, which is estimated from the aforementioned Rankine cycle simulation model at the whole range of engine loads examined. As revealed, there is an increase of the required working medium mass flow rate with engine load, for both working media. The estimated value for R245ca varies from 688.06 kg/h to 2381.98 kg/h and it is higher than the corresponding value for steam by a factor of ~15. The previous major difference is due to the significantly higher latent heat of steam compared to that of the organic medium R245ca. This, in conjunction with the lower vaporization pressure used when steam is the working medium, results to significantly lower pumping power losses. However in absolute values the corresponding power gain is not significant. Moreover it should be stated that due to the high mass flow rate required when the working medium is R245ca, a turbine expander is used, contrary to what happens when the working medium is water, where a piston expander seems to be preferable.

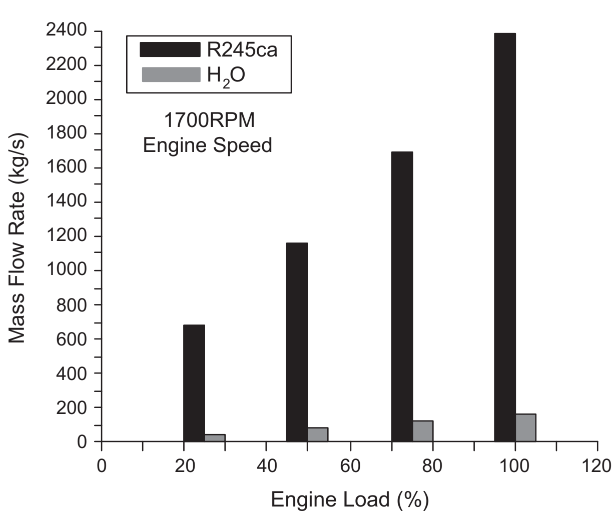

Fig. 5. Variation of the optimum mass flow rate value of steam and R245ca vs engine load at 1700 rpm.

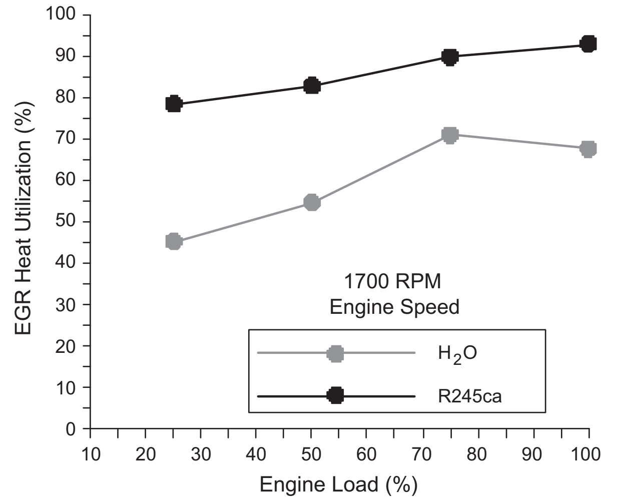

Fig. 6. Percentage of the EGR heat recovered from Rankine cycle operating with steam or R245ca vs engine load at 1700 rpm.

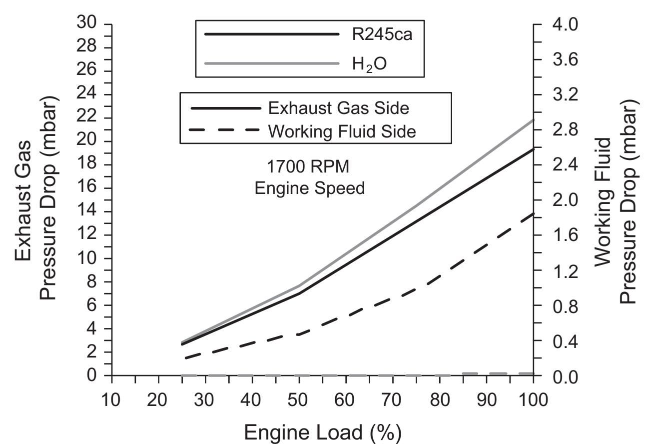

Fig. 7. Pressure drop at hot side and cold side of exhaust heat exchanger using either steam or R245ca as Rankine cycle working media vs engine load at 1700 rpm.

4.4. Extracted heat from the EGR cooler

In the configuration examined in the present study, part of the exhaust gas heat contained in the EGR loop is absorbed for heating up the Rankine cycle working medium. As witnessed in Fig. 6, the percentage of the EGR heat extracted by the organic Rankine cycle is higher compared to the corresponding one when using steam as working fluid. This occurs because for the organic medium the temperature level of the cold stream at the EGR heat exchanger is lower compared to the one for steam, enhancing thus heat absorption from the EGR stream. In Fig. 6 it is also revealed that the percentage of recovered EGR heat increases with engine load. The only exception to the previous observation is the operation of the steam Rankine cycle at 100% load, where there is a slight decrease of the percentage value of extracted EGR heat because of the high temperature values of superheated steam.

4.5. Effect on pressure drop

The pressure drop in the exhaust gas heat exchanger is a crucial parameter for the proper operation of a diesel truck engine equipped with a Rankine cycle system. As shown in Fig. 7, the impact of the exhaust gas heat exchanger on engine backpressure

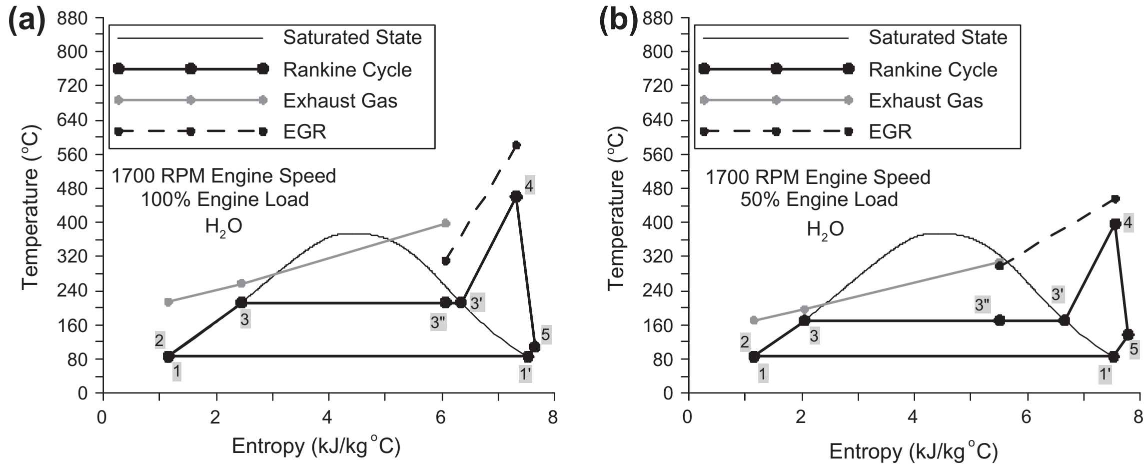

Fig. 8. Optimum steam rankine cycle and variation of exhaust gas and EGR temperature for 100% and 50% engine loads at 1700 rpm.

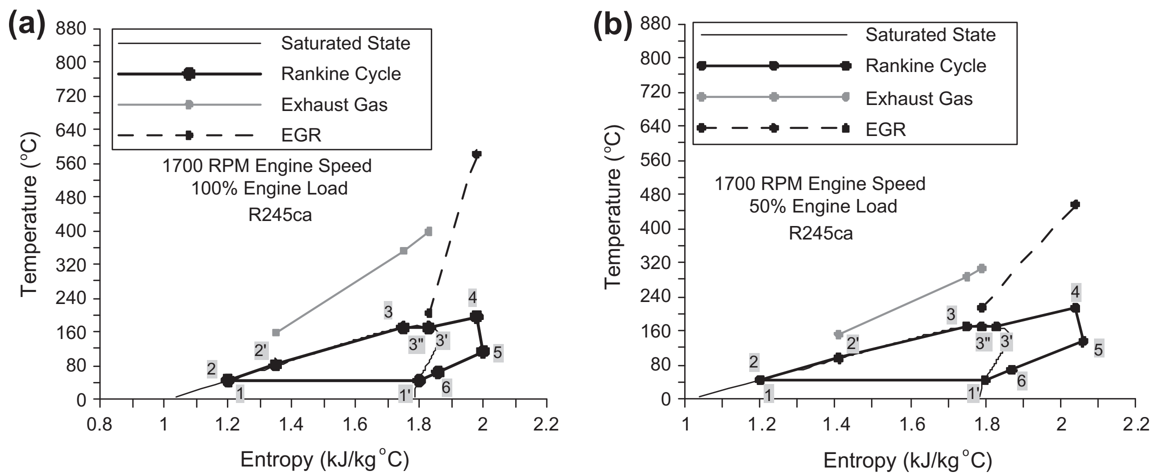

Fig. 9. Optimum organic rankine cycle and variation of exhaust gas and EGR temperature for 100% and 50% engine loads at 1700 rpm.

is rather limited since its maximum value is equal to ~22 mbar (when using water as working fluid). As for the cold side of the exhaust gas heat exchanger (working fluid side), the maximum pressure drop is very low and equal to ~1.83 mbar (when R245ca is the working fluid) and practically zero when using steam due to the considerably lower mass flow rate of working medium. Therefore, the assumption of the analysis that the pressure of the working medium is practically constant during its heating process is valid.

4.6. Effect on exhaust and EGR temperature-optimum Rankine cycle

In Fig. 8a and b, the optimum steam Rankine cycle is depicted on a temperature-entropy chart at full and part engine load. The saturated curves for both liquid and vapor are included in these figures. In these figures are also shown the temperatures of the exhaust gas during heat recovery at the main exhaust gas and EGR gas streams.

As witnessed, from these figures, the extracted heat from the EGR gas covers fully steam superheating. In addition, the high EGR gas temperature favors the superheating process of steam. It is also revealed that the required heat for evaporation is covered partially by the EGR gas heat and this is more obvious at part engine load (Fig. 8b). Another conclusion derived from the same figure is that the major part of heat extracted from the main exhaust gas stream covers the vaporization of water. Furthermore, the pinch point requirement is safely satisfied for the entire heating process of Rankine cycle. As observed in Fig. 8a and b, the working medium at the expander exit is superheated steam at both engine loads (50% and 100%). Due to the fact that the temperature of the EGR stream leaving the Rankine cycle system is greater than 290 °C, an additional EGR cooler is required to further reduce EGR gas temperature.

As far as the organic Rankine cycle is concerned, in Fig. 9a and b are shown the temperature-entropy charts for the cases of 50% and 100% of full engine load respectively. Similarly to what has already been observed for the steam Rankine cycle, in Fig. 9a and b is shown that the extracted EGR heat covers fully the superheating process of R245ca. At part engine load, a major part of the evaporation is also covered by EGR heat. The preheating process of the organic medium R245ca presents higher demands for heat compared to evaporation and superheating. Therefore, the major part of the absorbed heat from the main exhaust gas stream covers the preheating process of R245ca.

As observed from Fig. 9a and b, there is a significant drop of EGR temperature through the corresponding Rankine cycle heat exchanger at all engine loads for the organic medium. The EGR temperature drop is also greater compared to the previous case of the steam Rankine cycle, due to the superior thermodynamic characteristics of the organic medium. Moreover, it is revealed that the

heat recuperator reduces the rejected heat to the ambience which is favorable for the cooling system of the engine and at the same time supports the preheating process of R245ca improving Rankine cycle efficiency.

Taking into account the fact that engine load affects the Rankine cycle operation, it seems challenging to optimize the proposed system configuration under transient conditions. Specifically, the inertia of the examined system is the major obstacle for the transient operation. The exhaust gas heat exchangers, the condenser and the expander will delay to correspond in a possible variation of the exhaust gas mass flow and temperature. To overcome these difficulties, a heat recovery technology which is more appropriate for the transient operation of diesel engines is the electrical turbocompounding which reduces significantly the turbo lag [20].

5. Conclusion

In the present work a parametric investigation has been conducted to examine the potential of using a Rankine cycle system to recover exhaust gas heat from a diesel truck engine. The conducted theoretical investigation was based on a simulation model for the Rankine cycle developed by the present research group. For the evaluation of the cycle, two working media have been comparatively examined: water/steam and R245ca. From the analysis conducted it was revealed that the Organic Rankine cycle improves overall bsfc from 10.2% to 8.4% as engine load increases from 25% to 100% of engine load. On the other hand, when steam is used as the working fluid, bsfc improvement increases considerably with engine load and ranges from 6.1% up to 7.5% (for the same range of engine load variation).

The increase of Rankine cycle high pressure improves thermodynamic efficiency and increases cycle power output, although the generated power from the steam Rankine cycle starts to deteriorate after a certain high pressure value. The total heat extracted from the exhaust and EGR gas is reduced with the increase of cycle peak pressure, however the operation of the Rankine cycle at high pressure values is favorable for the engine cooling system since rejected heat to the ambience is reduced.

The maximum power generated by Rankine cycle system at full engine load is 34 kW when using R245ca and 30 kW when using steam as working medium for the present application. Rankine cycle with R245ca as working fluid presents almost the same optimum peak pressure value at all engine loads. On the other hand, the optimum high pressure value of steam Rankine cycle increases with engine load.

Organic Rankine cycle operates with 15 times higher mass flow rate of working fluid compared to the corresponding one when using steam. The optimum value of mass flow rate for both steam and organic media increases with engine load.

As far as the heat transfer is concerned, the low temperature values of the organic Rankine cycle favor heat extraction from the EGR gas. The impact of the exhaust gas heat exchanger on engine backpressure is limited. The pressure drop at working fluid side is also negligible. The preheating process of the organic medium R245ca absorbs the major part of the recovered exhaust gas heat. Superheating and partially evaporation of R245ca is covered by the extracted EGR heat.

On the other hand, water vaporization requires the greatest heat amount compared to preheating and superheating. The recuperated EGR heat covers fully steam superheating and partially water vaporization especially at 50% engine load. A supplementary EGR cooler is required, when the working fluid of Rankine cycle is steam.

Concluding, the installation of an organic Rankine cycle system on a truck diesel engine appears to be favorable since it improves significantly the total efficiency of the system. Nevertheless there exists one problem that should be considered for, the increased cooling demands of the combined system. Additional heat rejection from the Rankine cycle should be considered in future analysis, especially due to packaging restrictions for real truck applications.

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